HOMS & SOMS Opto-Mechanical Design

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Summary:

This document summarizes requirements, design, analysis, and test of the XTOD Soft and Hard X-ray mirror opto-mechanical system.

Change History Log

<table>
<thead>
<tr>
<th>Rev Number</th>
<th>Revision Date</th>
<th>Sections Affected</th>
<th>Description of Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>000</td>
<td>2008/12/16</td>
<td>All</td>
<td>Initial version</td>
</tr>
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</table>
DISCLAIMER

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1 Introduction

This document describes the requirements, design, analysis, and test of the HOMS and SOMS opto-mechanical systems. The primary function of the opto-mechanical system is to direct the FEL beam from the output of the undulator to various experiment stations, while separating the FEL beam from spontaneous radiation. This function must be performed within the requirements for FEL pointing, centering, and beam quality.

1.1 General Requirements

Requirements for the SOMS and HOMS mirror surfaces are described in LCLS-PRD-1.5-004 and 1.5-.005, respectively, and vacuum requirements are contained in LCLS-PRD-1.5-002. Engineering requirements that tier down to the opto-mechanical design are contained in this document, along with the description of a design that meets these requirements.

The HOMS and SOMS are covered in this single document because they closely parallel each other. A single document facilitates comparison while eliminating redundancy. ESD 1.5-125, XTOD Offset Mirror System Alignment, is a companion document that describes the origin of some of the beam pointing requirements described here.

1.2 Beam Path Layout

The Front End Enclosure (FEE), which contains the hard (2-8 keV) and soft (0.8-2 keV) x-ray mirrors, is shown in relation to the near and far hall experiment stations in Fig. 1-1. The soft x-ray mirrors deliver the beam to near experiment hutches, which are about 30 m away from the last soft x-ray mirror. The hard x-ray mirrors deliver the beam to far experiment hutches, which are about 300 m away. The long propagation path from mirrors to hutches has implications for pointing stability requirements.

An enlarged plan view of the FEE is shown in Figure 1-2. The soft x-rays reflect off M1S and M2S on their way to a branch line at M3S1/M3S2. The position and angle of mirror M3S1/M3S2 determines which of two paths the low energy beam will take to near experiment hutches #1 and #2. The first soft x-ray mirror is moved out the FEL beam path to switch to hard X-ray operation. This is illustrated more clearly in Figure 1-3. The hard x-ray beam reflects off the first hard x-ray mirror (M1H), and passes behind the second soft x-ray mirror, on its way to the second hard x-ray mirror (M2H).

The X-ray beam is pointed & centered in the horizontal plane. This requires two remotely-controlled degrees-of-freedom for each mirror. The beam location in the vertical is controlled by two factors: mirror assembly tolerances, and survey installation tolerances. These tolerances are described in Section 2.4.
Design Requirements

The SOMS and HOMS requirements documents PRD 1.5-004 and PRD 1.5-005 leave considerable room for definition of detailed design requirements. The intent of this section is to capture the spirit of these PRD’s, while deriving design requirements in enough detail to define a design. The design is described in Section 3, and measured performance of the design is compared to requirements in Section 4.

2.1 Mirror Figure

The mirror PRD’s state that the mirror must be flat enough to limit the beam divergence change from reflections. The intent is to preserve the FEL output intensity profile and phase. To limit the anamorphic power (i.e., spherical curvature in one axis) imposed by reflections, the divergence change from each reflection is limited to <10%. This requirement is imposed on the mirror surface exposed to the FEL beam, defined in ESD 1.5-119 and ESD 1.5-121 and summarized in Figure 2-1. The sagittal and tangential axis directions are indicated by arrows.

Spherical figure error in the tangential axis produces a uniform divergence change (spherical, parabolic and ellipsoidal figure are equivalent for the small curvature of interest here). Peter Stefan constructed Figure 2-2 to illustrate how beam divergence $\psi$ depends on mirror curvature $R$, beam diameter $w_{\text{beam}}$, and the mirror incidence angle $\alpha$. Numerical values for $\psi$ and $w_{\text{beam}}$ derived by Peter are in Appendix 2.A, and summarized in Table 2-1 for worst-case figure requirements. The value of $\alpha$ becomes $\sin(\alpha)$ for the sagittal direction, with $\alpha = \pi/2$. The curvature corresponding to 10% divergence increase is shown in Table 2-1, as calculated by the expression in Figure 2-2.
Figure 2-1. The cross hatched area shows dimensions that must meet the figure requirements.

For comparing engineering analysis and measurement, it is useful to convert curvature to peak-to-valley error relative to a straight line across the surface. The peak-to-valley error, $\Delta$, is related to curvature $R$, and the mirror illuminated length $2L_f$ (shown in Figure 2-1), by an expression derived in Appendix 2.B:

$$\Delta = \frac{L_f^2}{2R},$$

where the cross hatched areas in Figure 2-1 have length $2L_f$. The mirror tilt angle $\alpha$ requires the figure be maintained over much longer length along the tangential axis than the sagittal. The maximum figure error $\Delta$ is listed in the last column of Table 2-1 for each mirror.

The figure requirements in Table 2-1 are parsed into a budget for mounting, gravity, coating, thermal, and manufacturing contributions. The budget, shown in Table 2.2., is the result of iterative design analysis and test, described in Section 4. The peak to valley figure requirements correspond to a single reflecting surface. Because the SOMS beam line has three reflecting surfaces and HOMS two, the error from each mirror should be added in some manner. Multiplying by the number of mirrors $N$ may be overly pessimistic because the errors may not be correlated. Multiplying by the square root of $N$ is appropriate if the errors are completely uncorrelated, but there is no particular reason to think they are. It is possible the errors from multiple mirrors could be less than that of a single mirror. Given the uncertainty in how errors might add for multiple mirrors, we ignore this factor in this document while keeping the potential impact in mind.
Fig. 2-2: Illustration of beam divergence angle change due to mirror radius of curvature R.

Table 2-1: Requirements flow down for mirror cylindrical figure error

<table>
<thead>
<tr>
<th>Mirror Axis</th>
<th>Limit, μm</th>
<th>(W_{\text{beam}, \text{mm}}) (fwhm)</th>
<th>(\psi_0, \mu) (fwhm)</th>
<th>Radius, km</th>
<th>2(L_f), mm</th>
<th>Peak to valley, nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOMS Tangential</td>
<td>13.9</td>
<td>0.47</td>
<td>3.6</td>
<td>188</td>
<td>175</td>
<td>20</td>
</tr>
<tr>
<td>SOMS Sagittal</td>
<td>1570</td>
<td>0.47</td>
<td>3.6</td>
<td>3</td>
<td>10.0</td>
<td>5</td>
</tr>
<tr>
<td>HOMS Tangential</td>
<td>1.3</td>
<td>0.19</td>
<td>1.1</td>
<td>2657</td>
<td>444</td>
<td>9</td>
</tr>
<tr>
<td>HOMS Sagittal</td>
<td>1570</td>
<td>0.19</td>
<td>1.1</td>
<td>3</td>
<td>15.0</td>
<td>8</td>
</tr>
</tbody>
</table>

Table 2-2: Figure budget for HOMS and SOMS mirrors

<table>
<thead>
<tr>
<th>Mirror Flatness Requirement</th>
<th>Figure error budget, nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axis</td>
<td>&quot;Best Fit&quot; Peak to Valley, nm</td>
</tr>
<tr>
<td>HOMS/SOMS Sagittal</td>
<td>&gt;~ 3</td>
</tr>
<tr>
<td>SOMS Tangential</td>
<td>&gt;188</td>
</tr>
<tr>
<td>HOMS Tangential</td>
<td>&gt;2600</td>
</tr>
</tbody>
</table>
2.2 Mirror Pointing & Centering Requirements

2.2.1 Centering Resolution and Stability

The x-ray beam is aligned to the center of each mirror by translating the mirror normal to the beam path. The translation range must overlap the installation tolerances achievable by precision survey into a network, described in Section 2.4. Hence, the translation range used during mirror centering is listed as \(<\pm 1\text{ mm}\), with the two exceptions described next.

The translation range for M1S must exceed 6 mm to clear the beam path when hard x-rays are delivered to HOMS mirrors. The translation stage used to switch from M3S1 to M3S2 during low energy x-ray operation when must exceed 9.5 mm to pass the beam between mirrors.

If the x-ray beam is to be moved with 100 micron resolution along the mirror face, the translation resolution must be 100 microns divided by the mirror use angle. The resolution that results is listed in Table 2-3 for each mirror. The translation range and resolution requirements in Table 2-3 are easily met by commercial devices, built to a custom form factor.

Table 2-3: Translation and rotation range and resolution requirements.

<table>
<thead>
<tr>
<th>REQUIREMENTS</th>
<th>Remote Controlled Degrees of Freedom</th>
<th>SOMS</th>
<th>HOMS</th>
</tr>
</thead>
<tbody>
<tr>
<td>X Translation</td>
<td></td>
<td>M1S</td>
<td>M2S</td>
</tr>
<tr>
<td>Adjustment Range, mm</td>
<td>&gt;6</td>
<td>&gt;1</td>
<td>&gt;10</td>
</tr>
<tr>
<td>Nominal use angle, mr</td>
<td>13.9</td>
<td></td>
<td>1.3</td>
</tr>
<tr>
<td>Resolution, microns</td>
<td>1.4</td>
<td></td>
<td>0.13</td>
</tr>
<tr>
<td>Y Rotation</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Adjustment Range, +/-mr</td>
<td>3</td>
<td></td>
<td>1.2</td>
</tr>
<tr>
<td>Resolution, nanoradians</td>
<td>900</td>
<td></td>
<td>85</td>
</tr>
</tbody>
</table>

2.2.2 Pointing Resolution Requirements

The mirrors point the x-ray beam to downstream objects by rotating about the y (vertical) axis. The pointing range, defined in ESD 1.5-125 (XTOD Offset Mirror System Alignment), is limited to ensure the beam will always terminate on a safe aperture.

Rotation resolution must be fine enough to step the beam back into position at the experiment station if it drifts. Step increments should be less than 10% of the beam diameter, full width half maximum, to move the beam smoothly. Thus, the rotation resolution must be less than \(0.1W_{\text{exp}}/(2L)\), where \(W_{\text{exp}}\) is the beam width at the experiment station (not at the mirror). The distance \(L\) is measured from the last mirror to the experiment station, as shown in Figure 2-3. The rotation step required is determined by the smallest expected beam diameter. Peter Stefan suggests the smallest diameter is \(W_{\text{exp}} = 0.54\text{ mm}\) for SOMS (2 kV x-rays) and \(W_{\text{exp}} = 0.51\text{ mm}\) for HOMS (8.3 kV x-rays). The rotation resolution that results is listed in Table 2-3. The angle resolution required by SOMS is not particularly challenging, but the HOMS resolution is a challenge.
2.2.3 Pointing Stability Requirements

The x-ray beam position must be held at the experiment station with no noticeable change throughout the experiment duration, which could be weeks long. Hence, pointing stability should hold the beam within 10% of the diameter for extended periods. The stability requirements that result are listed in Table 2-3. The requirement is particularly challenging for HOMS.

Threats to pointing stability include amplified ground vibration, slow ground deformation, ambient temperature change, heating by the x-ray beam, and changing mechanical loads on the mirror pointing mechanism. The pointing mechanism is described in Section 3, and its performance in Section 4.

2.4 Assembly & Precision Survey Installation Tolerances

Installation errors are divided into three categories:

1. Survey network accuracy.
2. Chamber installation into the local network.
3. Mirror installation relative to chamber survey features.

Steps taken to control the error contribution from each category are described below. The tightest requirements on installation tolerances are determined by:

- The limited travel range of the mirror pointing cam,
- Beam pointing error in the vertical direction.

2.4.1 Network Error

The survey network that extends from the undulator to the FEE must be accurate enough to direct the FEL beam through 3 mm and 5 mm diameter collimator apertures. To do this, the network error must be ≤ 2.5 mm over 100 m length. Experience at SLAC and the LLNL has demonstrated that a carefully planned and well-maintained network can be accurate for hundreds of meters within +/- 0.3 mm, and verified by statistical means to 3σ confidence. It is assumed that the mirror systems will be installed in a network held to +/- 0.3 mm tolerance.

2.4.2 Chamber installation into the network

There are a total of six survey tooling holes located on the flanges at each end of the mirror vacuum chambers. Two of the holes on one flange, and one on the other, will be used to fix the chamber in six degrees of freedom (three translations and three rotations). One possible choice is shown in Figure 2-4, with coordinates and tolerances listed for all five mirror assemblies. A local coordinate system has been defined, somewhat arbitrarily, with the z origin at the reflection point of mirror M1S, but otherwise aligned to the incoming FEL beam. The other coordinates listed are relative to this origin, and position each mirror at its intended location and angle relative to MS1.
Figure 2-4: Example of survey points and coordinates for SOMS & HOMS mirror chambers. Target 2 tolerances (in black) are relative to the network, and those in red are relative to target 2.

Target 2 will be adjusted into its x, y, and z coordinates within a tolerance of +/- 0.3 mm relative to the network. Target 5 will be adjusted in x and y, with a tolerance of +/- 0.1 mm relative to target 2. Target 1 will be adjusted in y with a 0.1 mm tolerance relative to target 2. Installing targets 1 and 5 relative to target 2 (rather than relative to the network) minimizes the mirror roll error about axis z, and yaw error about axis y.

It is very important to verify all five mirror assemblies are installed relative to each other consistent with the tolerances listed in Figure 2-4. Once installation is complete, redundant survey data will be collected for all five mirror assemblies and surrounding network points. The survey data is used calculate an adjustment fit for the five mirror assemblies into the network. The adjustment fit provides an ellipsoid of installation uncertainty around each chamber survey point. The magnitude ellipsoid determines whether installation tolerances have been met within a 3σ confidence level. If tolerances are not met, the mirror assemblies will be moved by an appropriate amount, and the adjustment fit will be repeated.

2.4.3 Mirror installation error relative to chamber fiducials

Component stack up between the mirror’s reflective surface and chamber survey fiducials contributes to installation error. To mitigate this contribution, the chamber has a port that views the mirror face. Viewing the mirror face relative to survey fiducials with a laser tracker quantifies installation roll and yaw errors, as illustrated with Table 2-4.

The measured error is mitigated by modifying the nominal installation coordinates listed in Figure 2-4. The tolerance for measuring mirror installation relative to chamber fiducials is listed in Table 2-4. Error sources include fabrication errors associated with the mirror, the mount, and mount locating features machined into the vacuum chamber. The X, Y, & Z position tolerances for a point on the mirror’s reflective surface can be rather loose with no impact, and are set to +/- 0.25 mm (10 mils).

The mirror performance is relatively insensitive to pitch angle θx. The 2 mr in Table 2-4 can be met by holding machining tolerances on the mirror, mount, and chamber.

The mirror yaw angle tolerance θ y is held to +/-0.2 mr to limit the cam pointing range consumed by this error. This tolerance is not likely to be met by holding machining tolerances from the mirror face, through the mount, and to the chamber fiducials. Hence, inspection survey is likely to result in coordinate offsets to compensate. The roll angle tolerance in Table 2-4 reflects the need to constrain the vertical location of the x-ray beam on its way to the experiment station.
Table 2-4. Tolerance allocated to the mirror surface relative to the survey fiducials.

<table>
<thead>
<tr>
<th>Mirror Installation Tolerances, +/- (3o)</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Mirror</td>
<td>X, mm</td>
</tr>
<tr>
<td>SOMS &amp; HOMS</td>
<td>0.25</td>
</tr>
</tbody>
</table>

Impact of roll errors. Mirror roll errors (rotation about the z axis) induce vertical displacement of the X-ray beam from its intended location at the experiment station. The vertical displacement is the product of the reflected beam angle, propagation distance to the experiment station, and the roll error. To ensure the vertical displacement is within 1 mm of its intended location at the experiment station, the roll error should be < 1 mr for both HOMS and SOMS.

Impact of yaw errors. Yaw installation errors consume pointing cam range. The total range of the cam is +/- 3 mr for SOMS and +/- 1.2 mr for HOMS.

Network error is not a significant consumer of the available cam stroke. If network uncertainty is +/- 0.3 mm over a 12 m path between mirror isolation slabs, then the yaw uncertainty is only +/- 0.3 mm/12 m = +/- 0.25 μr.

The yaw installation error of the mirror tangential axes relative to each other is the primary consumer of cam stroke. The HOMS chambers are about 0.5 m in length. If 20% of the cam is to be dedicated to installation error, the mirror tangential axis must be installed relative to each other within about 0.3 milli-radians. This equates to +/- 1.2 mr * 0.5 m * 20% = +/- 0.1 mm displacement relative to the nominal installation coordinate. The adjustment fit analysis performed after initial survey installation is the primary means of ensuring this requirement is met.
Appendix 2.A Peter Stefan's Derivation of Mirror Radius of Curvature Requirement

FEL WIDTH DIV PLUS MIR LENGTH _HOMS_XMCD
2007/6/18  Peter M. Stefan
This worksheet uses the original version of 2006/6/29, with Richard Bionta's FEL beam width formulation from LCLS-TN-00-3, with the latest electron beam size data from Heinze-Dieter Nuhn's LCLS database. That version also does the arithmetic to obtain the beam footprint on the mirror, including the chin-guard and beam jitter. This version uses the HOMS M2H location for the worst-case beam spot sizes, etc., and a lower glancing angle for the HOMS mirrors, to conform to the present geometry anticipated.

Definitions:

\( m_e = 9.103826 \times 10^{-31} \text{ kg} \quad \text{eV} := 1.60217653 \times 10^{-19} \text{ joule} \quad \text{keV} := 10^3 \text{ eV} \quad \text{GeV} := 10^9 \text{ eV} \)

\( \mu\text{rad} := 10^{-6} \text{ rad} \quad \text{mrad} := 10^{-3} \text{ rad} \)

For this worksheet, origin in \( z \) is taken at the downstream end of the undulator.

Preliminaries:

\( \lambda_U = 30 \text{ nm} \quad K_U = 3.500 \)

Obtain the linac electron beam energy as a function of the undulator fundamental output wavelength:

\[
E_e(\lambda) = m_e c^2 \left( 1 + \frac{K_U^2}{2 \lambda^2} \right)^{1/2} \lambda_U \quad E_e(0.15 \text{ nm}) = 13.640 \text{ GeV} \quad E_e(1.5 \text{ nm}) = 4.313 \text{ GeV}
\]

Assume a linear variation of electron beam size from 4.313 GeV to 13.640 GeV, given rms beam sizes at these extremes:

\[
\sigma_{e\text{min}} = 36 \mu\text{m} \quad \sigma_{e\text{max}} = 40 \mu\text{m} \quad E_{e\lambda\text{min}} = E_e(0.15 \text{ nm}) \quad E_{e\lambda\text{max}} = E_e(1.5 \text{ nm})
\]

\[
\sigma_e(E_e) = \frac{\sigma_{e\text{max}} - \sigma_{e\text{min}}}{E_{e\lambda\text{max}} - E_{e\lambda\text{min}}} (E_e - E_{e\lambda\text{max}}) + \sigma_{e\text{max}}
\]

\[
\sigma_e(E_{e\lambda\text{min}}) = 36.000 \mu\text{m} \quad \sigma_e(E_{e\lambda\text{max}}) = 49.000 \mu\text{m}
\]

Obtain the beam width and divergence:

\[
w_0(\lambda) = \sqrt{2} \sigma_e(E_e(\lambda)) \quad w_0(0.15 \text{ nm}) = 50.912 \mu\text{m}
\]

\[
L_{\text{Rayleigh}}(\lambda) := \frac{\pi w_0(\lambda)^2}{\lambda} = z_0(\lambda) = L_{\text{Rayleigh}}(\lambda)
\]

\[
w(z, \lambda) = \sqrt{\left( \frac{z - z_0(\lambda)}{\lambda} \right)^2 + w_0(\lambda)^2}
\]

\[
w_{\text{FWHM}}(z, \lambda) = \sqrt{2 \ln(2)} \cdot w(z, \lambda)
\]

\[
w_{\text{FWHM}}(\lambda) := \frac{8 \ln(2)}{w_{\text{FWHM}}(z_0(\lambda), \lambda)} \cdot \frac{\lambda}{4 \pi}
\]
Collect the required data:

\[ \lambda = 0.15 \text{ nm} \quad z = 104.8 \text{ m} \]

\[ L_{\text{Rayleigh}}(\lambda) = 54.287 \text{ m} \quad w_{\text{FWHM}}(z_0, \lambda) = 59.944 \mu\text{m} \quad w_{\text{FWHM}}(\lambda) = 1.104 \times 10^{-6} \]

\[
\begin{bmatrix}
\lambda & L_{\text{Rayleigh}} & w_{0,\text{FWHM}} & w_{\text{FWHM}} \\
(\text{nm}) & (\text{m}) & (\mu\text{m}) & (\mu\text{rad})
\end{bmatrix}
\]

\[
\begin{array}{cccc}
1.5 & 10.057 & 81.590 & 8.113 \\
0.62 & 21.129 & 76.030 & 3.598 \\
0.15 & 54.287 & 59.944 & 1.104
\end{array}
\]

Add the arithmetic to calculate the mirror lengths:

\[ z_{\text{low}} = 105.4 \text{ m} \quad z_{\text{high}} = 104.8 \text{ m} \quad \text{(centers of downstream mirrors in each pair)} \]

\[ w_{50}(z, \lambda) = \frac{5 w_{\text{FWHM}}(z, \lambda)}{z \sqrt{2 \ln(2)}} \]

\[ \alpha_{\text{low}} = 13.9 \mu\text{rad} \quad \alpha_{\text{high}} = 1.324 \mu\text{rad} \quad \Delta_{\text{chin guard}} = 50 \mu\text{m} \quad \sigma_{\text{jitter}} = 0.25 \mu\text{rad} \]

\[ L_{\text{mir}}(z, \lambda, \alpha) = \frac{w_{50}(z, \lambda) + \Delta_{\text{chin guard}} + 2 \sigma_{\text{jitter}} z}{\alpha} \quad \text{(Jitter consideration is not so conservative. Assumes only 1\sigma excursion in each direction.)} \]

\[ L_{\text{Mlow}} = L_{\text{mir}}(z, 1.5 \text{ nm}, \alpha_{\text{low}}) \quad L_{\text{Mlow}^\prime} = L_{\text{mir}}(z, 0.62 \text{ nm}, \alpha_{\text{low}}) \]

\[ L_{\text{Mhigh}} = L_{\text{mir}}(z, 0.62 \text{ nm}, \alpha_{\text{high}}) \quad L_{\text{Mhigh}^\prime} = L_{\text{mir}}(z, 0.15 \text{ nm}, \alpha_{\text{high}}) \]

\[ \text{Spot}(z, \lambda, \alpha) = \frac{w_{50}(z, \lambda)}{\alpha} \quad \text{(Just the beam spot!)} \]

\[ \text{Spot}_{\text{Mlow}} = \text{Spot}(z, 1.5 \text{ nm}, \alpha_{\text{low}}) \quad \text{Spot}_{\text{Mlow}^\prime} = \text{Spot}(z, 0.62 \text{ nm}, \alpha_{\text{low}}) \]

\[ \text{Spot}_{\text{Mhigh}} = \text{Spot}(z, 0.62 \text{ nm}, \alpha_{\text{high}}) \quad \text{Spot}_{\text{Mhigh}^\prime} = \text{Spot}(z, 0.15 \text{ nm}, \alpha_{\text{high}}) \]
\[
\begin{align*}
\text{width}_{\text{Mlow}} &= \sigma_{\text{Low}} \cdot 1.5 \text{ nm} \\
\text{width}_{\text{Mlow}}' &= \sigma_{\text{Low}} \cdot 0.62 \text{ nm} \\
\text{width}_{\text{Mhigh}} &= \sigma_{\text{High}} \cdot 0.62 \text{ nm} \\
\text{width}_{\text{Mhigh}}' &= \sigma_{\text{High}} \cdot 0.15 \text{ nm}
\end{align*}
\]

\[
\begin{align*}
\text{fwhm\_width}_{\text{Mlow}} &= \text{fwhm}_{\text{Low}} \cdot 1.5 \text{ nm} \\
\text{fwhm\_width}_{\text{Mlow}}' &= \text{fwhm}_{\text{Low}} \cdot 0.62 \text{ nm} \\
\text{fwhm\_width}_{\text{Mhigh}} &= \text{fwhm}_{\text{High}} \cdot 0.62 \text{ nm} \\
\text{fwhm\_width}_{\text{Mhigh}}' &= \text{fwhm}_{\text{High}} \cdot 0.15 \text{ nm}
\end{align*}
\]

Collect the results:

<table>
<thead>
<tr>
<th>System</th>
<th>(Z_{\text{mir}})</th>
<th>(\lambda)</th>
<th>(\lambda_{\text{Rayleigh}})</th>
<th>(w_{\text{fwhm}_{\text{Zmir}}})</th>
<th>(\sigma_{\text{width}})</th>
<th>(\text{Spot})</th>
<th>(\text{Length})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low</td>
<td>106.4</td>
<td>13.9</td>
<td>1.5</td>
<td>0.948</td>
<td>2.013</td>
<td>144.856</td>
<td>152.280</td>
</tr>
<tr>
<td>Low</td>
<td>106.4</td>
<td>13.9</td>
<td>0.62</td>
<td>0.465</td>
<td>0.988</td>
<td>71.056</td>
<td>76.490</td>
</tr>
<tr>
<td>High</td>
<td>104.8</td>
<td>1.324</td>
<td>0.62</td>
<td>0.459</td>
<td>0.976</td>
<td>736.868</td>
<td>814.209</td>
</tr>
<tr>
<td>High</td>
<td>104.8</td>
<td>1.324</td>
<td>0.15</td>
<td>0.186</td>
<td>0.394</td>
<td>297.565</td>
<td>375.007</td>
</tr>
</tbody>
</table>

\[
\begin{align*}
\lambda_{\text{Rayleigh}} &= w_{0 \text{FWHM}} / w_{\text{fwhm}} \\
(\text{nm}) &= 0.62 10.057 81.590 8.113 \\
(\mu\text{m}) &= 76.030 3.598 \\
(\mu\text{rad}) &= 59.944 1.104
\end{align*}
\]
Appendix 2.B Relating Peak to Valley Figure to Radius of Curvature

Let arc OC be the mirror neutral axis, with OC = L, where \( L \) is half the mirror length.

The radius \( R \) of neutral axis arc OA = AC, and angle \( OAC = \alpha \).

Then \( L = \frac{R \alpha}{2} \) or \( \alpha = \frac{L}{R} \).

From similar triangles, angle DCB also = \( \alpha \).

The angle between cord line DC and a line tangent to OC ranges between 0 and \( \alpha \) along the path of OC. Then for small \( \alpha \), the average angle \( \approx \frac{\alpha}{2} \). In this case length DO = OB.

The deformation \( \delta \) we seek - length DO. Then

\[
\delta = R(1 - \cos(\alpha)) = R \frac{\alpha^2}{2} - \frac{R(L/R)^2}{2} = \frac{L^2}{2R}.
\]

For \( L = 10/2 \) mm and \( R_{\text{avg}} = 1 \) km, \( \delta_{\text{avg}} = 12.5 \) mm

For \( L = 175/2 \) mm and \( R_{\text{avg}} = 50 \) km, \( \delta_{\text{avg}} = 76 \) mm
3 Design Approach

The SOMS & HOMS opto-mechanical system consists of the five assemblies illustrated in Figure 3-1. The assemblies include the:

- support pedestal & floor anchors
- installation alignment adjustments
- pointing & centering devices
- mirror & mount assembly
- vacuum chamber.

These assemblies are briefly introduced in the following sections. Because the HOMS & SOMS systems are very similar, they can be largely described without distinguishing between them, with distinctions made as necessary.

3.1 Support Pedestal & Floor Anchors

Mirror system installation begins by surveying locations for threaded fasteners on the FEE concrete floor. Six 5/8” diameter Kwik bolt-TZ (Hilti) mechanical anchors are sunk into the floor. Nuts and spherical washers are used to level the pedestal on the anchors, with a nominal gap of 2.5” between the bottom of the pedestal and the floor. After the pedestal is leveled and each anchor secured with top nuts, the gap between the pedestal and the floor is filled with grout. The grout is important for limiting vibration amplification of the overall assembly, but is discounted in analysis of seismic safety.

The pedestal is made of 14” diameter carbon steel tube with 0.5” thick wall. The nominal height and weight of each mass center is shown in Figure 3-2. Normal loads are listed in the table, along with seismic loads from 0.57 g in the horizontal direction. Bending moments from the horizontal loads and moment arm are also listed. The seismic safety factor for interaction forces on the anchors is five, as calculated by the Profis Hilti analysis program.
Figure 3-2. Loads and lever arms used to calculate the seismic safety factor of the anchors.

3.2 Installation Alignment Plates

The adjustable installation alignment plates are shown in Figure 3-2. The first plate above the pedestal provides tip/tilt angle adjustment, plus vertical (y) adjustment. Three 5/8” diameter Mic-posi leveling devices (Harbinger Industries) are located in the ¾” thick steel tip/tilt plate. The angle resolution of these adjusters is within a few milli-radians. The vertical height range is +/- 6 mm, and the resolution is well within +/- 0.3 mm. The Mic-posi devices are lockable after the three adjustments are complete.

A second ¾” adjustable steel plate rests on top of the tip/tilt plate for translation adjustment. The bolt holes used to fasten the second plate to that below are over sized, accommodating +/- 5 mm translation in the x and z axis. Manually driven push screws located on all four sides control the translations. Translation resolution is within +/- 0.3 mm. While this second plate provides some z rotation adjustment, it is not meant to be the final z rotation installation alignment.

The z rotation installation alignment adjustment is done using a ½” thick steel plate that rests on top of the translation plate. The rotation plate is pinned at its center to the center of the translation plate for precise rotation control. Rotation is driven clock wise or counter clockwise by a pair of drive screws located near the plate edge. Resolution of this adjustment is fine enough to locate the chamber within the acquisition range of the cam used in mirror pointing alignment.

3.3. Pointing & Centering

Pointing & centering is facilitated by two sub-assemblies: a translation slide, and the chamber rotation sub-assembly. The general arrangement is shown in Figure 3-3, and an enlarged view of the translation and rotation components is shown in Figure 3-4.

Components for remote rotation and translation are all located outside the vacuum chamber. This provides the simplest physical arrangement of motion devices, and the most compact vacuum chamber. Vibration and thermal drift, described in Section 4, are more easily studied and controlled with alignment components in air rather than in vacuum.
3.3.1 Translation Slide Assembly

The translation slide is a custom 6” wide x 10” long stage built by American Linear Slide. The customized feature is its width, which the manufacturer has increased to 10” to provide a wider footprint for chamber rotation components. The slide carriage has cross roller bearings for stiffness, and provides +/- 0.5” of travel, with adjustable mechanical limit switches at each end. The bottom of the slide is bolted to the top installation alignment plate. The slide is constructed of cast iron. During alignment the X-ray beam will centered first, followed by precision pointing. Thus, rotation errors inherent in slide motion (see Section 4.2.2) do not interfere with precision pointing.

The slide carriage is driven by a 10 TPI ball screw, preloaded to eliminate free play. Motion resolution is provided by a 100:1 harmonic drive (for both SOMS and HOMS) coupled to a 1000 step/revolution motor. Hence, the motion resolution of 1 micro-inch, or 0.025 microns/step, well exceeds the required centering resolution.

Figure 3-3. View of the HOMS vacuum chamber with pointing & centering mechanisms.

Figure 3-4. Enlarged view of the chamber translation & rotation components.
3.3.2 Chamber Rotation Sub-assembly

The vacuum chamber rotation axis is defined by a six bladed flexure spindle, with blades arranged for high stiffness in all degrees of freedom except y rotation. The spindle axis is nominally located at the center of each mirror face, which decouples centering adjustments from pointing. Only the SOMS mirrors M3S1 and M3S2 are offset from the mirror face, by about 5 mm, which couples rotation adjustments into beam centering. However, for these X-ray beam incidence angles, the coupling is small enough to ignore.

Chamber rotation is driven by a circular offset cam. Roller bearings are pressed onto an offset cylinder machined into a shaft, and a 2 axis flexure is clamped around the bearings. As the cam rotates, the offset pushes (or pulls) on the vacuum chamber through the two axis flexure. The cam undergoes sinusoidal displacement in the vertical and cosine displacement in the horizontal direction. The flexure decouples the cam vertical motion from the chamber to provide only horizontal motion. The second flexing axis accommodates the small chamber displacement along z axis as the chamber rotates.

Bearings pressed onto each end of the cam shaft are captured by bearing blocks, which are bolted to the top of the translation slide carriage. Timken Thin Super Precision angular contact bearings are used throughout. The bearing races are slightly deformed as the bearing block and flexure are clamped, eliminating clearance between the bearings and races. This approach eliminates backlash, but the clamping force must be applied with care (feeling the motion as bolt force is applied) to avoid overly rough motion between the balls and races.

Rotation resolution is determined by the factors listed in Table 3-1. The theoretical resolution shown is for the cosθ=1 position: when the cam offset is in the vertical orientation. The measured resolution is described in Section 4.2.1.
Table 3-1 Theoretical rotation resolution of cam driven mirror pointing

<table>
<thead>
<tr>
<th>Mirror System</th>
<th>Cam Offset, mm</th>
<th>Lever arm, mm</th>
<th>Step Motor, steps/rev.</th>
<th>Gear Reduction</th>
<th>Range, mn(r/.)</th>
<th>Resolution, nr</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOMS</td>
<td>0.45</td>
<td>150</td>
<td>1000</td>
<td>100</td>
<td>3.0</td>
<td>188.5</td>
</tr>
<tr>
<td>HOMS</td>
<td>0.3</td>
<td>250</td>
<td>1000</td>
<td>2500</td>
<td>1.20</td>
<td>3.0</td>
</tr>
</tbody>
</table>

3.4 Mirror Assembly

The mirror assembly consists of the following components and sub-assemblies:
- A single crystal silicon mirror
- An Invar 36 mirror mount
- Three coil spring assemblies, two horizontal clip springs, and one axial clip spring
- The B₄C chin guard
- The mirror bender (HOMS only)

Most features of the HOMS and SOMS sub-assemblies are similar. The HOMS sub-assembly is shown in Figure 3-5. The HOMS bender is not shown, and will be discussed separately.

Figure 3-5. Exploded view of a HOMS mirror sub-assembly.

3.4.1 Single crystal silicon mirror

The mirror is supported at six points of contact, or constraints, with the mount. The six constraints are labeled in Figure 3-5. There are three constraints in the y direction, two in the x direction, and one along the z direction. The mirror is pressed against each constraint by mounting springs.
The mounting features machined into the mirror are designed to minimize the strain field from mounting spring forces. The strategy is to minimize the volume of mirror material in strain, and keep it as far from the mirror reflective surface as reasonably possible. To this end, the three holes for the vertical (y) holding force are counter-bored to reduce the pinched volume, and lower it below the zone used by the x-ray beam. Vertical slots machined for the x constraint springs minimize the pinched volume, and are located near the back of the mirror.

The z holding force and its corresponding constraint are handled differently. Instead of minimizing the pinched volume, they can be nominally located along the neutral (z) axis of the mirror to minimize bending. The force of a constraining spring in the z direction spreads uniformly over the cross section. Upon final assembly of the mirror on the mount, the z force and constraint can be offset along the horizontal plane. This induces a mild bending moment to precisely tune the mirror tangential figure. For SOMS mirrors, this is a manual operation, completed in front of an interferometer. The HOMS assembly, a manual operation may not by itself result in the required flatness. Hence, for HOMS a remotely-controlled mirror bender allows fine figure adjustment during operation.

3.4.2 Mirror mount

The mount has three support pads machined into it as y constraints. The pads form a triangular pattern to support the mirror stably against gravity. Coil spring assemblies (a spring and an attachment post) apply a few pounds of y holding force to the mirror. The attachment post passes through holes in the mirror, and threads into tapped holes in the mirror mount y constraint pads.

Flexures machined around the mirror mount pads allow the mirror to move slightly relative to the mount and relax friction force between the mirror and pad. The flexures accommodate differential thermal expansion between the mirrors and mount, and are also in play when bending the HOMS mirror. An expanded view of a flexure for the HOMS mount is shown in Figure 3-5. The flexure blades are stiff enough (about 1 lb/micron) that ground vibration does not excite a resonant frequency, but soft enough that friction at the mirror and mount interface boundary is relieved. The flexure geometry is the result of iterative numerical simulations of thermal expansion effects, and the action of the HOMS mirror bender.

The mount material is invar 36, which is a good match to single crystal silicon thermal expansion. Published values for invar 36, which vary with suppliers, and depend on annealing, are around 1.2x10^-6/C (http://www.edfagan.com/2006/technical-assistance/#). Published values for bulk single crystal silicon also vary, and are around 2.4x10^-6. (Proc. Phys. Soc., LXXV, 5, R. Birss and R. Horne, 1959, p. 793). The finished invar mount is flashed with a few angstroms of nickel and then 100 angstroms of gold to prevent surface oxidation.

Hard stops bolted onto the mount react against the horizontal spring forces, and constrain the mirror in the x and z directions. The loads from these stops and springs are applied in mirror y mid-plane to minimize unintended bending moments.

The mirror mount has three support pads machined onto its bottom surface to secure it to the vacuum chamber: two at one end of the mount, and one at the other. Upon installation into the chamber, the two mounting bolts on one end are tightly secured. The third bolt on the other end of the mount applies a light spring force. The light force allows the mirror mount subassembly to slide relative to the chamber during pump down or thermal expansion. This sliding action prevents chamber deformation from propagating into the mount, and thus the mirror figure.

3.4.3 Chin guard

The chin guard blocks x-rays that would otherwise strike the end of the mirror at near normal incidence to avoid localized mirror heating. The blocked x-rays are scattered and attenuated by a
piece of $B_4C$ mounted near the side of the mirror that faces the incident beam, as shown in Figure 3-6. Heat absorbed by the $B_4C$ block is radiated to the vacuum chamber and mirror, with little consequence to either.

The $B_4C$ block must be carefully aligned to the mirror surface to avoid excessive protrusion into the FEL x-ray beam. After alignment the SOMS chin guard should stand beyond the mirror surface by less than 520 microns. The stand off of the HOMS chin guard should be less than 25 microns to shadow only the first 25 mm of mirror surface.

![Figure 3-6. $B_4C$ chin guard used to protect the end of the mirror from normal-incidence x-rays, shown with assembly fixtures.](image)

Chin guard alignment to the mirror surface is accomplished with the aid of a gage block assembly tool. Chin guard alignment requires rotating it about the z axis, translating in the x axis, and rotating about the y axis. The gage block tool is first set against the unused surface of the mirror to provide a precision reference surface. The chin $B_4C$ chin guard is then pressed against the tool surface (by hand or using spring loaded plungers). The gage block has a step machined into it to align the chin block surface relative to the mirror surface. Adjustment screws on the chin block mount are carefully secured to lock it in place, and the alignment tool is removed. Because the HOMS offset is so small, it is advisable to leave the alignment tool in place up to final installation of the mirror assembly in the facility.

### 3.4.4 Mounting springs

Three stainless steel coil spring assemblies hold the mirror against the mount vertical (y) stops. The coil springs are pinned at one end to the top of a post. The other end of the post screws into holes tapped into the mirror mount flexure. The coil spring slips around the post and is compressed against a counter-bored surface machined into the mirror. The force on the mirror from each compressed spring is adjustable, but less 3 lb.

Two stainless steel leaf springs fit into slots machined into each end of the mirror to hold it against the mounts two x-mount constraint posts. The leaf springs compress against the mirror slot as they are bolted to the mirror mount. They are 0.015” thick stainless steel, and each provides about 2 lb of force once compressed against the mirror.

For the SOMS mirror subsystem, the sixth constraint is provided by a stainless steel leaf spring that bolts along the z axis of the mirror mount. The spring holds the mirror against the z
constraint at the other end of the mount, and applies about 3 lb of force along the mirror z axis. Its location can be varied to introduce a bending moment of either sign into the mirror, and thus vary the tangential mirror figure. For the HOMS mirror subsystem, the sixth constraint is provided by the mirror bender assembly described next.

### 3.4.5 HOMS bender

The HOMS tangential figure requirement may be too tight to be met by adjusting the figure on an interferometer prior to installation. The risk of failing to meet the figure requirement is reduced if the figure can be adjusted in-situ during operation, using the x-ray beam intensity profile as a metric. Remote control of tangential figure requires an adjustable bending moment be applied during operation. The adjustable bending moment is applied with the bender mechanism shown in Figure 3-7.

The mechanism consists of a pivot flexure clamped into a manually adjustable block, labeled manual force adjustment in Figure 3-7. The pivot flexure mid-section is clamped to a forcing element that presses near the back of the mirror to induce bending. During assembly on the interferometer, the manual adjustment block is rotated to vary the bending force the pivot flexure induces on the mirror, leaving the mirror as flat as detectable by interferometer measurement error.

The mirror is fabricated with 150 nm of concave sphere to ensure only compressive force on the mirror is needed to adjust it flat on the interferometer. The manual adjuster is then locked to hold the manually applied bending moment constant. Finite element analysis of the mounted mirror assembly shows about 6°-lb moment is required to remove 200 nm of spherical figure.

Once the mirror assembly is installed into the vacuum chamber, a spring is attached to the element that applies bending force to the mirror. The bending force can now be varied during operation by extending or compressing the spring. The spring is designed to produce +/- 6 lb force from +/- 4 mm slide travel, which induces +/- 200 nm from the nominally flat condition of the installed mirror. The spring constant is low enough that mirror figure is quite insensitive to temperature variations of all components.

The spring force acting on the forcing element is remotely controlled by a linear slide located outside the vacuum chamber. The slide is connected to the spring by a rigid shaft welded onto a mini-conflat flange & bellows. The vacuum force of the bellows acts on the slide lead screw, not on the coil spring, to decouple it from the spring force applied to the mirror. Bending resolution, and hence figure resolution, is determined by the linear slide lead screw and stepper motor resolution. The lead screw has 1 mm/turn pitch and 1000 step/revolution motor, which results in a theoretical figure modification resolution of 0.05 nm per step.

![Figure 3-7 View of the HOMS mirror bender components.](image)

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3.5 Vacuum Chamber

The vacuum chamber is a 6” diameter stainless steel tube with non-rotating Conflat flanges welded onto each end. The SOMS chamber is about 12” long, and the HOMS chamber is about 24” long. Sections of welded bellows with 4” ID are bolted to each of the Conflat flanges. Each bellows has one rotating and one fixed Conflat flange. During operation the bellows are subject to a few mm transverse offset, which results in a few pounds of reaction force on each end of the chamber. The chamber rotation and translation devices easily overcome these forces due to high gear reduction.

Stainless steel mirror mount support brackets, visible in Figure 3-8, are welded into the chamber interior. After welding, the bracket surfaces are each finish machined and a notch is added to receive locating pins pressed into the mirror mount. The notch is tolerated relative to the survey tooling features in the chamber Conflat flanges. Three survey tooling features are machined into the flanges: one on top, and two on the sides of each flange.

A cradle structure, welded to the vacuum chamber exterior, provides an interface to the chamber flexure spindle. The two axis flexure that is attached to the rotating cam assembly, and drives chamber rotation, bolts onto this cradle.

The vacuum chambers have 2.75” Conflat flanges near each end. These ports are used to inspect location of the installed mirror relative to the chamber fiducials with a laser tracker. They also accommodate any unanticipated future use.

3.6 Instrumentation

The electrical components of a mirror assembly are listed in Table 3-2. The translation, rotation, and bender mechanisms use 1000 step/rev. 5 wire Oriental motors.

The translation and bender stages have a pair of limit switches to indicate end-of-travel before encountering a hard stop. The rotation mechanism does not have limit switches, and has no hard stop position: it can rotate continuously.

LVDT’s are used as absolute position indicators for the translation, rotation, and bender mechanisms. The location and range of the LVDT’s is shown in Figure 3-8.

Table 3-2. Electrical components required for a mirror assembly. The black dots indicate the number of wires per connector.

<table>
<thead>
<tr>
<th>Connector List</th>
<th>Motor</th>
<th>Limit</th>
<th>LVDT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Translation slide:</td>
<td>☒</td>
<td>☒</td>
<td>☒</td>
</tr>
<tr>
<td>Rotation cam:</td>
<td>☒</td>
<td>☒</td>
<td>☒</td>
</tr>
<tr>
<td>HOMS bender:</td>
<td>☒</td>
<td>☒</td>
<td>☒</td>
</tr>
</tbody>
</table>
Figure 3-8. Location of LVDT's used to measure device position (bender LVDT not shown).
The connector list in Table 3-2 is an accurate depiction for the M2S, M1H, and M2H mirror system. However, the translation slides for M1S and M3S1/M3S2 require additional switches that interface with the Machine Protection System. The layout and wiring schematics for these switches are shown in Figure 3-9. These switch configurations were established by Peter Stefan’s assessment of what is required to ensure adequate machine protection from a misdirected laser beam. The dimensions shown illustrate travel distances between switch contact points.

Figure 3-9. Wiring schematic for translation slide limit switches, illustrating normally opened or closed configuration. Dotted lines indicate the switches that interface with the Machine Protection System.
4 Design Performance

This section summarizes analysis and measurements performed to verify the opto-mechanical design meets its requirements. Section 4.1 describes various contributors to mirror figure error, and the HOMS mirror bender performance in controlling tangential power.

Section 4.2 describes measurements conducted to verify the performance of pointing & centering mechanisms, including motion resolution, backlash, and vibration stability.

Section 4.3 describes how pointing stability has been measured as a function of room temperature variation. These measurements are particularly challenging, as pointing stability must be measured with 10 nr scale resolution over many hours. The section closes by describing how pointing stability requirements can be met by surrounding the mirrors in an enclosure whose interior temperature is stabilized within ± 0.01ºC.

4.1 Mirror Figure

Four contributions to figure error were considered:
- mounting forces,
- thermal deformation,
- gravity induced deformation,
- and coating stress.

Finite element calculations are presented in Sections 4.1.1 through 4.1.4 for each contribution. The calculations for mounting, thermal, and gravity-induced figure change have been verified by experiment using HOMS and SOMS prototype mirrors and mounts. However, the resolution of the interferometer used to measure figure change is limited to about +/- 2 nm by turbulence and/or vibration, and in some cases the figure errors predicted by calculation are less than the interferometer resolution. In this case the measurements show no measurable figure change.

The total deformation is summarized for both SOMS & HOMS mirrors in Section 4.1.5. The performance of the HOMS mirror bender in controlling tangential figure is also described in Section 4.1.5.

4.1.1 Mounting Deformation

The deformation caused by 3 lb. spring forces acting on the mirrors has been calculated by finite element analysis of the mirror and mount. The general arrangement of the mirror and mount were described in Figure 3-4. For the SOMS mirror, there are six fixed spring forces and constraints. Three coil springs hold the mirror to the flexure pads, and two leaf springs hold the mirror to posts at the back of the mount. A variable spring force in the z direction holds the mirror to the z constraint post. For HOMS mirror analysis, the variable z force was set to zero because the influence of the HOMS mirror bender was analyzed separately. The action of the flexures machined around the y-mount constraints is included in the analysis of both mirrors. The mirror is assumed to be bonded to these flexures in the analysis, which is a worst-case for mirror deformation.

Figure 4-1 shows the calculated deformation from mounting the SOMS mirror mount along the 175 mm tangential path (see Figure 2-1). The displacement shown is normal to the mirror face, with tilt removed. The calculated displacement is less than 1 nm. The calculation includes all six 3 lb forces and constraints, but figure error is primarily due to the three vertical forces. Counter-boring the vertical mounting holes deep into the mirror has reduced the volume of mirror material strained by the vertical springs, and the resulting displacement on the mirror surface.
Figure 4-1. Deformation induced at the SOMS mirror surface from six 3 lb. spring forces.

Figure 4-2 shows the calculated deformation from the HOMS mirror x and y mounting forces, along the 430 mm tangential path (Figure 2-1). The figure change shown is the component normal to the mirror face, with tilt removed. The holes counter-bored in the HOMS mirror have been moved close to the mirror ends to better accommodate the action of the mirror bender. This seems to have changed the character of the curve compared to SOMS mounting deformation.

To validate mounting deformation calculations, prototype SOMS and HOMS mirrors were fabricated from fused silica. Prototype mounts, with flexures machined into them, were made of Invar 36. The spring forces were applied and deformation measured on a Zygo 12” interferometer. The surface change measured from the spring forces was often smaller than the interferometer measurement resolution, which is limited by turbulence and vibration to +/- 2 nm. This noise floor is evaluated by taking data sets without changing the mirror figure, and subtracting the data sets from each other. The spring forces must be applied carefully to the HOMS mirror. It is much more sensitive to how they are applied due to its longer aspect ratio.
4.1.2 Thermal Deformation

The heat flux on the mirror during operation consists of spontaneous radiation and coherent FEL x-rays. Peter Stefan estimated the worst-case average heat load as 260 mW spontaneous, and 10 mW coherent x-rays, respectively. The coherent heat flux (in W/cm²) follows from the beam diameters listed in Table 2-1, and the tangential length shown in Figure 2-1. The spontaneous heat flux is calculated assuming 5 mm diameter collimators define the width along the tangential length (less the portion shielded by the chin guard). The reduction in spontaneous radiation that occurs from successive reflections is ignored, consistent with a worst case analysis.

Thermal equilibrium is calculated using a finite element model that includes grey body radiation of the mirror and mount to the vacuum chamber, as well as radiation coupling between the mirror and mount. The mirror reflective surface emissivity is assumed to be 0.2, and 0.6 for the five etched surfaces. The mount emissivity is assumed to be 0.8, consistent with a diffuse gold coating. Thermal conductivities used are 130 W/m/K for single crystal silicon, and 10.4 W/m/K for Invar 36. The chamber temperature is fixed at 300 K for the calculations.

The SOMS mirror and mount equilibrium temperatures are shown in Figure 4-3 and 4-4 respectively. The mirror temperature is very uniform, and its gradient is not evident in the color iso-contour plot. The temperature plotted along the tangential axis illustrates the distribution. The bulk temperature rise is less than a degree for HOMS and SOMS, and slightly smaller for HOMS because of its larger volume. The SOMS temperature distribution is asymmetric because the chin guard shadow covers a larger fractional length than it does for HOMS.

The mirror and mount temperature distributions are input to a finite element calculation of thermal deformation. To isolate thermal deformation, all mounting spring forces were set to zero. The temperature gradients tend to bend the mirror face convex, but mounting flexure reaction forces also influence the shape. The interface between the mirror and mount flexure pads is assumed bonded to account for differential expansion between the mirror and mount. The mirror surface deformation will be smaller if slippage occurs at this interface.

Figures 4-5 and 4-5 are iso-contour views of total thermal displacement for HOMS and SOMS, along with a plot of the displacement component normal to the mirror face. The iso-contour gradients show the total displacement is primarily along the axial (z) direction, but a deformation component normal to the mirror surface is also evident. Blue represents zero displacement, and red is maximum.

The deformation is concave for SOMS: opposite the convex shape expected for a freely expanding mirror. The concave curvature is the result of reaction forces between the mirror and mounting flexures. The reaction forces are less for HOMS because the temperature rise is smaller, and the curvature has the convex shape of a freely expanding mirror. The displacement direction reverses near the HOMS mirror ends as the result of an abrupt transition to free ends. This transition effect is not evident for SOMS because 175 mm tangential length does not approach the free ends.
Figure 4-3. Temperature distribution for the SOMS mirror and mount. The temperature plot along the tangential axis of the mirror face illustrates the gradient.

Figure 4-4. Temperature distribution for the HOMS mirror and mount. The temperature plot along the tangential axis of the mirror face illustrates the gradient.

Figure 4-5. Total thermal displacement of the SOMS mirror. The mirror normal displacement is plotted along the tangential axis.
4.1.3 Gravity Deformation

The effect of gravity on the mirror and mount displacement is shown in Figure 4-7 and 4-8. The color iso-contour images show the total deformation due to gravity. The SOMS mirror is relatively stiff compared with the mount, and does not sag significantly. The HOMS mirror sag is comparable to that of its mount. The interface between the mirror and mount is un-bonded for these calculations, because gravity acts on the mirror before surface friction applies.

The SOMS figure error is much less than 1 nm, and the HOMS figure error is only slightly larger. The abrupt transitions may be due to boundary conditions used in the calculation, but magnitudes are small enough they do not warrant further investigation into perfecting the calculation. Gravity-induced errors are difficult to validate experimentally because there is no practical method of turning gravity off.
Figure 4-8 HOMS mirror & mount total displacement is shown in color. The mirror surface normal displacement due to gravity is plotted along the tangential axis.

4.1.4 Coating Deformation

The coatings applied to HOMS & SOMS mirrors are in compression after deposition onto the substrate. Hence, the coating process induces mirror deformation. The coating stress can be deduced by measuring deflection of a thin sample substrate after deposition. For B\textsubscript{4}C coatings used on the SOMS mirror, the maximum stress is estimated (by Regina S. and Mike P.) to be $\sigma_c = 0.5$ GPa in a coating of thickness $t_c = 10$ nm. The stress in the SiC coating used on the HOMS mirror is expected to be much less than the B\textsubscript{4}C coating for SOMS, but for conservatism we assume the product $\sigma_c t_c$ for HOMS is one half that of SOMS.

Because the coating stress is assumed to be uniform over the entire 10 nm layer, the figure error that results can be estimated analytically if end effects are relatively small. Appendix 4.A shows how coating deformation $\delta$ scales with the product $\sigma_c t_c$, Young's modulus $E$, the mirror half width $c (= 25$ mm for SOMS and HOMS), and the mirror length $2L_f$ listed in Table 2.1. The peak to valley deformation calculated from this expression is 3.5 nm for SOMS, and 11 nm for HOMS. This contribution is plotted along with other contributions in the deformation summary listed in the next section.

The coating deformation calculated for the SOMS mirror was verified by measuring the figure of a full sized single crystal silicon blank before and after coating. The net deformation was primarily sphere, and the magnitude was consistent with the calculation described above.

4.1.5 Deformation Summary and the HOMS Bender

The deformation contributions described above are summarized in Figure 4-9 for SOMS and HOMS. The net sum is also shown. The net deformation is convex for both, with a maximum of 4.5 nm for SOMS, and 11.5 nm for HOMS. The requirement for SOMS tangential figure in Table 2-1 is 20 nm. Figure 4-9 implies this requirement can be met without active figure adjustment during operation. An interferometer with a transmission flat calibrated by 3-flat test should be capable of verifying the SOMS mirror assembly meets this requirement before it is installed in the chamber. If necessary, the SOMS tangential figure can be adjusted while on the interferometer by moving the $z$ axis mounting spring off the mirror neutral axis.

For HOMS, the net deformation predicted by analysis exceeds the figure requirement in Table 2-1 by more than a factor of two. This indicates the HOMS mirror requires figure adjustment after mounting. A 3 flat calibration of the interferometer may allow the figure to be verified within 10 nm over a 450 mm aperture. However, it is reasonable to question how well this figure can be held during installation and operation.
To mitigate risk, the HOMS mirror figure will be adjustable in-situ during operation. The need for adjustment will be determined by observing the change in beam diameter before and after reflection off the HOMS mirrors. The device that actively controls the HOMS mirror figure, introduced in Figure 3-7, is referred to as the bender.

Because the net figure from various contributions is calculated to be convex, it is reasonable to fabricate the HOMS mirror with concave pre-figure, and then bend it flat when the mirror is installed on the mount. Consistent with this approach, 150 nm of concave pre-figure has been incorporated into the HOMS mirror fabrication specification.

Figure 4-9. Various deformation contributions along the tangential axis are shown for SOMS and HOMS, along with the net total deformation.

The bender design has been demonstrated experimentally, and its performance compared with predictions from a finite element model of the mirror and mount. The prototype HOMS mirror assembly is an Invar 36 mount and fused silica mirror. The figure calculated by the finite element model for a 2.7 lb bending force acting on the mirror is shown as the blue line in Figure 4-10.

The change in figure after applying a bending force was measured with a 12 inch aperture interferometer. The interferometer can only view about 290 mm of the 450 mm tangential length at a time, but the data demonstrates the features of interest. The mirror figure change measured by the interferometer is shown as the red line. The measured deformation is within 10% of the calculated value, consistent with uncertainty in the applied bending force.

The departure from sphere predicted by the finite element model is shown in Figure 4-11. This curve is produced by subtracting sphere from the calculated blue line in Figure 4-10. The finite element analysis predicts bending the mirror results in slight aspheric curvature near the free ends.

The departure from sphere was measured by viewing half the mirror as shown by the cross hatched area in Figure 4-12, and subtracting sphere from the interferometer data. The measured figure is plotted with sphere removed. Any departure from sphere is hidden in the resolution noise of the interferometer data. The noise floor is limited by turbulence over large spatial scale, and vibration at smaller spatial scale. The overall noise uncertainty appears to be about +/- 1.5 nm.
Figure 4-10. The deformation predicted when a 2.7 lb force is applied to the HOMS prototype is shown as the blue line for the full length of the mirror. The deformation measured on the interferometer is shown on the right, as limited by the 12 inch interferometer field of view.

Figure 4-11. Distortion predicted by the finite element model from bending the mirror with 2.7 lb force.
4.1.6 Interferometer Calibration

The previous section describes how figure changes were measured by comparing interferograms before and after a force acts in or on the mirror. This is adequate for evaluating contributions to figure degradation after the mirror is fabricated. However, adjusting the mirror flat during final assembly requires an absolute flatness standard. This standard is typically created by calibrating the interferometer’s transmission flat with the aid of two other flats in a 3-flat test. In its usual form, the 3 flat test measure's flatness along the vertical and horizontal axis. The data often accompanies the flat shipped by the interferometer manufacturer.

However, there is a complication when performing the 3-flat test for 12” Zygo transmission flats. Calibration along a horizontal line (the X-ray mirror tangential axis) requires rotating one of the transmission flats about the horizontal axis. Zygo recommends a 12” transmission flat never be rotated about its horizontal axis. Reversing the weight puts epoxy bonds at three radial support tabs in tension, and could possibly tear them off the flat. To discourage rotating transmission flats about a horizontal axis, Zygo transmission flat cells are designed to prevent them from being mounted in this configuration. As a result, flats 12” and larger come from Zygo with a calibration along the vertical axis, but not along the horizontal axis.

This calibration problem was overcome by substituting one of the transmission flats in a 3-flat test with a reflection flat. The reflection flat mounting cell permits it to be rotated in all three axes, because its mounting cell does not use epoxy bonds. Presumably this is because reflection flat figure errors need not be as carefully controlled in its applications. The un-silvered side of the reflection flat was used, because the silvered side had too many blemishes visible on the interferogram.

The geometry and data processing algorithms for two transmission flats plus one reflection flat are shown in Appendix 4.B. The data set measured for the third configuration involves a sign change in the x coordinate. This is accommodated in data processing by reversing the sign of the x coordinates in this data file. The sign reversal is symbolized as F3(-x) in the Appendix.

The figure of the interferometer transmission flat measured by the three flat test described above is shown in Figure 4-13. The figure to the left compares the measured vertical figure with the
data provided by Zygo for this flat. The agreement is within the +/- 2 nm random noise floor from vibration and turbulence.

Figure 4-13. Results of the three flat test to calibrate the horizontal and vertical line of the 12” transmission flat.

The horizontal line calibration from the 3-flat test is shown in right hand figure. The figure’s magnitude and slope are similar in character to the vertical calibration. The size of the SOMS and HOMS mirrors are also shown for reference. The horizontal calibration will be subtracted from interferograms measured for the X-ray mirrors, to arrive at the “true” mirror figure. This subtraction will not result in a significant correction for SOMS because the transmission flat slope is ~linear over the SOMS length. However, the correction is more significant over the length of the HOMS mirror.

The first three configurations shown in Appendix 4.B allow the horizontal figure to be calculated for the three flats. Once the figure is known, there are two alternate configurations that allow the systematic error associated with reversing the reflection flat weight to be evaluated. These two alternate configurations are also shown in Appendix 4.B. Finite element analysis indicates that the error from weight reversal should be quite small about the horizontal axis. This was confirmed by taking data and using the algorithm shown in Appendix 4.B to evaluate Error(x). The curve indicating the magnitude of Error(x) is shown in Figure 4-13, plotted along with the horizontal line calibration. The data confirms the systematic error from reversing the reflection flat weight is small.
4.2 Pointing and Centering Motion Resolution

Motion resolution, linearity, hysteresis, and backlash have been measured for pointing and centering prototype mechanisms. The data is described below.

4.2.1 Pointing Motion

The horizontal displacement imparted by the cam to the chamber in X is given by

\[ X(n) = H \sin\left(\frac{2\pi n}{GS}\right), \]

where

- \( n \) = the number of motor steps in a command,
- \( H \) = the cam offset distance,
- \( G \) = the harmonic drive gear reduction,
- \( S \) = the steps per motor revolution.

For the HOMS prototype \( H = 0.5 \text{ mm} \), \( G = 2500 \), and \( S = 1000 \). The displacement measured by a Microstrain Differential Variable Reluctance Transformer (DVRT) transducer, plotted in Figure 4-14, shows the expected curvature for a half rotation of the cam.

![Figure 4-14. Displacement vs. motor steps for a cam half revolution.](image)

DVRT’s have the precision required to measure the micro-radian scale resolution required to point & center mirrors with pop-in monitors. However, they do not have the resolution required to verify that beam motion at the experiment station will be within a fraction of the beam diameter. We use capacitance sensors with nm scale position resolution to demonstrate that requirement. Capacitance sensors are not required during operation, but are essential for demonstrating pointing resolution of the design concept.

Capacitance sensors were used to investigate the curve in Figure 4-14 in higher detail near the origin, where motion resolution is at a minimum and the slope is at a maximum. Near the origin the sin function can be replaced by its first term expansion, \( X(n) = \frac{H2\pi n}{GS} \). The prototype had a 0.5 mm cam, and a HOMS vacuum chamber with 230 mm lever arm. Hence, the theoretical resolution, plotted as the black line in Figure 4-15, is 5.5 nm/step. The first cycle is over +/- 1000 steps in 100 step increments, followed by a second cycle over +/-100 steps in ten step increments.

The motion with ten step increments meets the HOMS pointing resolution requirement. The motion is linear, with little backlash upon reversal or lost motion over a full cycle. For increment commands less than a few steps, the motion becomes relatively unresponsive and inconsistent. As a result, the theoretical resolution of 5.5 nm is not achieved.
The cam shaft drives rotation of the flexure spindle (six blades for SOMS, twelve blades for HOMS) that supports the target chamber, as illustrated in Figure 4-16. The twelve bladed design provides higher stiffness to limit ground-vibration-induced pointing jitter. At full rotation, the reaction force by the rotation flexure blades on the cam is 25 lb. This is under the 100 lb maximum recommended by Timken for precision bearing motion, and the buckling safety factor for the cam flexure is 27. The cam flexure blade has a slight wedge machined into it to distribute stress more uniformly along its length, which limits the maximum stress to 6.5 ksi.

The stress that results in the HOMS rotation spindle and cam flexures at full rotation is shown in Figure 4-17. The twelve blades of the flexure spindle have steps machined into them at locations that limit flexure stress, along with the force required to rotate the chamber.
Resonant modes of the HOMS rotation assembly have been calculated. The fundamental mode, shown in Figure 4-18, has a resonant frequency of 90 Hz, and couples strongly into pointing. The spring constant measured for the cam assembly, $2.2 \times 10^5$ lb/in, was used in this calculation. The fundamental is calculated to be 95 Hz when the cam is infinitely stiff, indicating much of the compliance is in the cam and rotation spindle flexures. While calculating resonant modes is somewhat instructive, directly measuring ground-vibration-induced pointing jitter is far more useful. The pointing jitter measurements are described in Section 4.3.2 below.

Figure 4-16. Displacements for SOMS cam and rotation flexures.

Figure 4-17. Maximum von Mises stress in the HOMS cam and rotation flexures.
Figure 4-18. Vibration analysis shows that the HOMS assembly fundamental frequency is limited by the cam & bearing assembly.

4.2.2 Centering Motion

The translation slide ball screw is 10 TPI (threads/in), and driven by a 100:1 harmonic drive coupled to a 1000 step motor. The theoretical linear motion resolution is 25 nm/step, which far exceeds the requirement for centering the beam on the mirror.

The ability of the slide to hold a rotation angle as it moves was measured. The rotation change of the slide carriage was measured over a 0.5” range of motion, repeated ten times. The angle error from each motion is shown in Figure 4-19, starting with zero pointing error at position 0. The end of each movement is numbered sequentially, and the rotation error at that position was measured.

The rotation error from slide translations is significant. This behavior is not a problem for pointing and centering the mirrors using pop-ins because pointing tolerances are relatively loose. But clearly the slide must be held stationary when pointing the beam to experiment stations.
4.3 Pointing Stability

Two contributions to time-dependent pointing error are considered: pointing jitter (> 1 Hz) due to ground vibration amplification, and slow errors (< 1 Hz) driven by temperature change. Section 4.3.1 addresses the effects of thermal drift, and Section 4.3.2 addresses vibration.

4.3.1 Thermal Stability

The temperature history in a typical air conditioned room can be very dynamic. The temperature variation measured over a 36 hr period in one test laboratory is shown as the blue curve in Figure 4-20. The thermal spectra contains time periods from seconds to hours. The high frequency temperature fluctuations can be filtered out by enclosing the test volume with insulation. The red line in Figure 4-15 shows the same room air fluctuations when the volume is insulated with 2” of styrofoam. Periods shorter than a few minutes are filtered. However, periods greater than a few hours are quite effective at penetrating the insulation.

Figure 4-19. Change in rotation angle from translating the slide. Odd numbers indicate a move in one direction, and even indicate a move in the opposing direction.
The effect of temperature fluctuations on mirror pointing stability can be troublesome for SOMS, and catastrophic for HOMS. The pointing error measured for a SOMS prototype rotation assembly is shown in Figure 4-21. Assembly components with low thermal mass track rapid air temperature variations, giving rise to rapidly fluctuating pointing error. This is superimposed on errors induced by components with higher thermal mass, which cause slower pointing errors.

A second curve in Figure 4-21 shows how insulating the test enclosure eliminates fast pointing error, while the error driven by slow temperature fluctuation is unaffected. This data suggested a two-pronged approach:

1. insulate the volume of interest, and
2. regulate the air temperature inside the insulated volume.

Regulating air temperature inside the enclosure is practical because the thermal impedance of the insulation reduces the energy exchange rate.

Sensor Error: Accurately measuring nano-radian scale pointing error over many hours required use of new generation capacitance sensors from ALD Technologies. Long-term measurement stability was demonstrated by placing a pair of sensors in a stationary gauge block shown in Figure 4-22, and measuring the voltage change vs. time. The figure shows the sensor error measured over a seven day period, converted to an equivalent rotation error for the HOMS mirror assembly. These measurements were made with the sensors inside a temperature-controlled enclosure.
enclosure stable within +/- 0.01 C. The red line shows the temperature of the sensors. The sensor electronics were exposed to room temperature variations, which is the orange line. The black line is the sensor rotation error.

Figure 4-22. Pointing error measured using two ALD 8100 sensors in a stationary gage block, converted to an equivalent error for the HOMS rotation mechanism.

During the first two days, room temperature excursions exceeded the enclosure temperature set point, warming the sensors slightly. The rotation sensor error is higher during these two days. During the next five days the sensor temperature was controlled within 0.01 C, while and room temperature fluctuations were also smaller. Overall it seems the rotation error may scale with room temperature fluctuations with an error coefficient around 10 nano-radians/1ºC.

**Thermal Pointing Drift Measurements:** The experiment set up for measuring thermal pointing stability is shown in Figure 4-23. The mirror assembly was surrounded by 2” of styrofoam, with a re-circulating air cabinet cooler attached. A chiller provides water regulated within +/- 0.1ºC to the cabinet cooler heat exchanger. The insulation and cabinet cooler limit thermal excursions within the enclosure to a few tenths of a degree, while the outside room air fluctuations are ten times greater. The temperature inside the thermal enclosure, with the chiller on, is plotted along with outside air temperature in Figure 4-23. The insulation only extends to the bottom of the strut assembly because the focus of these measurements was on the adjustable mechanisms.
Figure 4-23. Experiment set up for measuring pointing thermal stability

**Mirror Pointing Mechanism:** Pointing stability measured over multiple days is shown for the mirror pointing mechanism in Figure 4-24. The sensor output, converted to rotation error for HOMS, is displayed on the right hand scale. The temperature variation inside the test enclosure during these measurements is displayed on the left hand side. The air temperature outside the enclosure varies by more than +/- 1°C during this test, imprinting itself on the inside temperature.

The pointing error dependence, \( \frac{d\theta}{dT} \), is evident from the rotation change per unit temperature change. The pointing error is highly correlated with temperature, with little random behavior. Overall, the thermal response of this mechanism corresponds to about 70 nr of angle change per 0.1°C temperature change.

Figure 4-24. Pointing error temperature dependence for the HOMS rotation mechanism.

**Translation Mechanism:** The pointing error induced by temperature change of the translation slide was measured in manner similar to the rotation assembly. Capacitance sensors measured rotation of the top carriage relative to the stationary bottom as temperature varied. The
results, shown in Figure 4-25, indicate the translation slide pointing error is more sensitive than that of the pointing mechanism. The derivative for the translation slide is \( \frac{d0}{dT} = 300 \text{ nrad per } 0.1^\circ \text{C} \) change. A temperature excursion of +/- 0.5°C was deliberately induced in the heat exchange water at about the middle of this data set. The error coefficient measured during the spike is consistent with that due to drifts in room air temperature.

![Figure 4-25. Pointing error temperature dependence for the translation stage used to center the mirrors.](image)

**Strut Plates:** The prototype mirror assembly tested in Figure 4-23 shows six alignment struts between the pedestal and the mirror motion mechanisms. The rotation stability measured between the bottom and top strut plate is shown in Figure 4-26. The pointing error that is correlated with temperature (\( \frac{d0}{dT} \)) is about 100 nrad per 0.1°C of temperature change. In addition to the correlated error, the strut assembly exhibits a significant drift uncorrelated with temperature. To eliminate the uncorrelated drift, the strut assembly was replaced by the slide and rotation plates described in Section 3.2. The new design has much lower correlated pointing error coefficient \( \frac{d0}{dT} \), and the uncorrelated pointing error is not measurable.

![Figure 4-26. Pointing error between top and bottom strut plates over a five day period.](image)
**Thermal Control** With a pointing error coefficient around 300 nr/0.1°C for the translation slide, the temperature around the mirror assembly must be stable within +/- 0.01°C to limit mirror pointing drift less than +/- 30 nrad.

The approach to controlling temperature around the five mirror assemblies is shown in Figure 4-27. A local thermal enclosure will be constructed around the two vibration isolation pads that support mirror assemblies. The temperature inside the enclosure will be held slightly higher than the maximum room temperature outside the enclosure. Temperature stability will be provided by controlling the power supplied to a resistive heat source inside the enclosure. The temperature will be measured and provided to a feedback algorithm that controls the heat dissipated by the source, ensuring it matches that conducted through the walls and into the floor.

The thermal control concept was tested by surrounding the prototype assembly shown in Figure 4-23 with six inches of Styrofoam insulation. The insulating enclosure completely covered the mirror assembly, and flexible insulation was laid inside on the concrete floor and around the pedestal. The heat flow path through the enclosure is shown schematically in Figure 4-28. Temperature inside the enclosure was measured within +/- 0.01°C using a thermistor and a Eurotherm 2404 controller. The controller provided power to a 3” muffin fan that dissipates up to 15 watts.

The temperature outside and inside the enclosure are shown in Figure 4-28. The controller limited variations inside the enclosure within +/- 0.01°C. As a result, the thermally induced pointing error of the translation stage, also shown in Figure 4-28, was limited to only +/- 30 nrad, which meets the pointing stability requirements for HOMS. This demonstrates the viability of controlling thermally-induced mirror pointing errors using local insulated enclosures.

![Figure 4-27. Insulated enclosures will be constructed around each of the two mirror assembly isolation pads to provide temperature control.](image-url)
Figure 4-28. Schematic of the prototype mirror assembly enclosure, and the heat flow path, are shown on the left. The temperatures inside and outside the enclosure are plotted on the right, along with the HOMS rotation error measured for the translation slide.

4.3.2 Amplified Ground Motion

Mirror pointing error induced by un-amplified ground motion is small compared to pointing stability requirements. However, if natural frequencies of the mirror support structure overlap the ground spectra, then resonances will amplify ground motion to produce significant pointing jitter. Resonances are avoided by making the mirror support structure stiff enough, where definition of “stiff enough” is described below.

The prototype assembly pedestal was 10” diameter, 0.13” wall carbon steel tube. After the rotation and translation assemblies were placed on the pedestal, scope traces from the capacitance sensor signals were captured to measure rotational jitter from amplified ground vibration. The measured amplitudes are shown in Figure 4-29.

Figure 4-29. Rotation jitter of prototype translation, strut, and rotation assemblies induced by ground motion amplified in the prototype pedestal.

Vibration Finite Element Analysis Finite element analysis of the pedestal and strut assembly was conducted to determine how the design should be modified. The first five calculated resonant frequencies are listed in Figure 4-30. The first two modes involve a significant amount of pedestal tube bending. The higher modes involve strut bending from the inertial force of the vacuum chamber and motion assemblies.
An improved design with a larger pedestal diameter and wall thickness is shown in Figure 4-31. The lowest calculated mode is limited to 46 Hz by vibration of the strut assembly. The strut assembly can be stiffened by adding redundant struts after installation alignment is complete. The first mode frequency increases to 78 Hz by adding two redundant struts to each of four sides.

The fact that the strut assembly is not inherently stiff, and the uncorrelated, thermally-induced pointing drifts described in Section 4.3.1, suggested the strut assembly should be replaced by a more robust installation alignment assembly. This led to the installation alignment plate design described in Section 3.2. Vibration analysis of the alignment plate design is shown in Figure 4-32.
The first two modes are at 83 Hz, and the mass participation factors show that they are limited by the pedestal stiffness. The total weight of this design is about 750 lb, roughly equivalent to the original design using the 10” pipe and adjustable struts. Hence, it is considerably stiffer, without increased weight.

**Vibration Measurements** While finite element analysis provides a reliable assessment of structure frequency response, the displacement amplitude is limited by damping, and is best assessed by measurement. The approach to estimating displacement amplification from spectral density measurements made with an accelerometer is described below.

The spectral density measured on the ground near the pedestal base is shown in Figure 4-33. The units are in mean square acceleration per unit frequency, because ground vibration is a random, time averaged quantity. The magnitudes measured in the two horizontal directions (not shown) are comparable, but slightly smaller. The measured spectrum is compared to a reference spectrum that represents a nominally quiet design environment. Overall, the data would indicate that the floor vibration is “nominal”, and not unusually noisy. Some of the peaks measured on the ground are due to resonances exited in the near-by prototype structure, and not necessarily representative of the “natural” ground spectra.

![Figure 4-32. Finite element analysis shows that the final design is much stiffer than the original prototype.](image)
Figure 4-33. Vertical spectrum measured at the floor of the prototype test assembly.

The mean square displacement in direction $x$ is related to the ground spectral density $S(f)$ by (Thomson, Theory of Vibration, 3rd ed.)

$$\langle x^2 \rangle = \int_0^{\infty} S(f) H(f) H^*(f) df,$$

where $H(f)$ is the prototype structure’s frequency response function, and $H^*$ is its complex conjugate. To get a feel for displacement scaling, assume the structure response is described by a net spring coefficient $k$ and damping coefficient $\xi$. Then

$$H(f) = \frac{1}{k\left[1-(f/f_n)^2 + i(2\xi f/f_n)\right]},$$

where $f_n$ is a particular resonant frequency of the structure. If the ground excitation spectrum $S(f)$ is relatively flat compared to the structure’s resonances, then the integrand is only significant near a resonance. In that case $S(f) = S(f_n)$ can be extracted, and the above integral evaluated to yield the displacement from a single resonant frequency:

$$\langle x^2 \rangle = \frac{\pi f_n S(f_n)}{(4\xi k^2)}.$$

As anticipated by Hooke’s law for a damped spring/mass, the rms displacement (vibration induced angle jitter) is proportional to the rms excitation amplitude, inversely proportional to the spring constant, and inversely proportional to the square root of the damping coefficient. The expression above shows why stiffening the structure is more effective at reducing vibration than adding special damping features.

In the approximation above, constants $f_n$, $k$, and $\xi$ divide out when taking ratios of $S(f)$ measured at different locations on the structure. Thus the relative magnitude of displacements, or displacement magnification factor, is given by

$$\frac{\langle x \rangle}{\langle x_{\text{ground}} \rangle} = \sqrt{\frac{S(f)}{S_{\text{ground}}(f)}}.$$
This expression allows the structure’s contribution to vibration amplification to be quantified from the values of $S(f)$ measured at different locations.

The above expression was used to quantify the amplification factor of the prototype mirror assembly. Figure 4-34 shows the horizontal component of spectral density measured on the floor near the prototype. The spectral density measured on top of the pedestal (at the bottom strut plate and labeled mid level), and on the top strut plate (labeled top level) are shown for comparison. The data indicates the fundamental frequency of the prototype is around 20 Hz (the finite element analysis predicted 25 Hz). From the above expression, the amplification factor $\langle x \rangle/\langle x_{\text{ground}} \rangle$ is about 100 from the floor to the lower strut plate, and 145 to the top strut plate.

The data in Figure 4-34 also shows resonant amplification in the 50 to 80 Hz range, which is also consistent with the finite element vibration analysis. The amplification factors are similar in magnitude to those around the fundamental frequency. The amplification factors calculated from spectral density measurements explain the vibration jitter measured on the prototype mirror assembly (plotted in Figure 4-29). The final design is over an order of magnitude stiffer than the prototype. Hence, the vibration jitter of the final design is lower by an order of magnitude, making it a negligible contribution to pointing error.

![Spectral density measured on the floor in a transverse direction compared to that on the bottom and top strut plates.](image)

Figure 4-34. Spectral density measured on the floor in a transverse direction compared to that on the bottom and top strut plates.
Appendix 4.A Coating Stress and Mirror Distortion

- Coating stress $\sigma_c$ induces mirror bending stress $\sigma_m$
  
  $$\sigma_m = \frac{Mc}{l} = 3\sigma_c \frac{t_c}{(2c)}$$
  
  at mirror surface, $= 3 \text{ kPa}$

- This stress induces strain $\varepsilon_m$
  
  $$\varepsilon_m = \frac{\sigma_m}{E}, \text{ where } E = 130 \text{ GPa}$$

- From geometry, $\varepsilon_m = \frac{c}{R}$
  
  Then $R = \frac{2c^2E}{3\sigma_c t_c}$
  
  $= 1075 \text{ km in both axis}$

- Induced figure error $\delta = \frac{L_i^2}{2R}$
  
  $\delta_{tan} = 7.2 \text{ nm for SOMS},$
  
  $= 23 \text{ nm for HOMS},$
  
  $\delta_{sag} = 0.01 \text{ nm for both}$
Appendix 4.B Data processing algorithms for calibrating transmission flat T1 and T2, and reflection flat R

- Change File3 x coordinate sign to convert T2(-x) \( \rightarrow \) T2(x) and R(-x) \( \rightarrow \) R(x), then solve:

\[
T1(x) = \frac{1}{3}[F1(x) + F2(x) - F3(-x)] \\
T2(x) = \frac{1}{3}[F1(x) - F2(x) + F3(-x)] \\
R(x) = \frac{1}{3}[-F1(x) + F2(x) + F3(-x)]
\]

- The figure error from “flipping” the RF can be evaluated

\[
File4(x) = T1(x) + [R(x) + Error(-x)] \\
File5(x) = T2(-x) + [R(x) + Error(x)]
\]

Check
\[
Error(x) = F4(-x) - T1(-x) - R(x) \\
\text{and double check} \\
Error(x) = F5(x) - T2(x) - R(x)
\]